

EFFECT OF DIFFERENT FUELS ON COMBUSTION AND EMISSION COMPONENTS OF CI ENGINES

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ABSTRACT

Combustion characteristics of different fuels (diesel, toluene, n-butanol, n-heptane and methane) of a direct injection diesel engine during combustion were investigated using computational fluid dynamics (CFD) simulation techniques. The relationship between fuel combustion reaction components and combustion temperature and turbulence rate was analyzed. The results show that the calculated value of the cylinder temperature verifies the correctness of the model. From the simulation results, it can be found that the total temperature in the n-heptane cylinder has the smallest NO_x and the largest turbulence rate. However, in the n-heptane cylinder, the NO_x and SO₂ reactive groups generated by the combustion reaction are the least. From the simulation results, it can be found that the different fuels have different reaction characteristics inside the engine, and the reaction products are caused by the physical and chemical properties contained therein. Due to the limitations of the simulation, we can compare the reaction characteristics and exhaust emission characteristics of other fuels in future experiments and simulation studies.

KEYWORDS: Combustion, CFD, Fuels, Direct Injection & Turbulence Rate

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INTRODUCTION

Diesel engine combustion characteristics and emission performance can be depend on by fuel styles in engines cylinder combustion, like fuel injection timing, fuel vapor, fuel mixed physics characteristics and engine structure like piston geometry and cylinder surface structure. And the chemical reaction kinetics of the working fluid in the cylinder is closely related to the physicochemical properties of the fuel. The combustion characteristics of the fuels with different types, different hydrocarbon components and different chemical compositions are the key to combustion. The composition changes are also different. So in the world, a lot of researchers were investigated the combustion performance of various fuels in diesel engine.

Biofuels are renewable fuels. In the evolution, secondary regeneration can be achieved to reduce environmental pollution. Therefore, biofuels are potential substitutes for petroleum fuels and can effectively address the energy waste crisis.

Alcohols such as ethanol and methanol, commonly referred as lower alcohols, are fuels used in conventional CI fuels mixed with conventional diesel fuels and are the most widely tested alcohols. However, fuel mixtures have some disadvantages, for example, their very low calorific value and high evaporation enthalpy, resulting in incomplete combustion of the fuel, which degrades engine performance and increases engine CO and HC emissions. In addition, ethanol and methanol have high resistance to self-ignition due to their low cetane number. In addition, due to its higher vapor pressure and lower flash point, more stringent safety precautions are needed in fuel handling and storage. In

addition, ethanol and methanol have poor lubricity and miscibility with conventional diesel, so alcohol fuel is used in CI engines. In order to avoid these incompatible shortcomings, the use of CI engines requires the search for other types of fuels as a blend of engines.

Recently, researchers have begun to focus on the use of higher alcohols (C4-C20) instead of diesel because their containment properties are close to that of diesel compared to the corresponding properties of lower alcohols. Higher alcohols therefore include butanol, pentanol, hexanol, heptanol, octanol, decyl alcohol and most phytol. Due to its higher calorific value, higher alcohol content, better mixing stability with diesel, higher density, higher viscosity, better lubricity and higher cetane number and higher the quality of the ignition, it shows good potential and characteristics in the operation of the diesel engine. Higher alcohols with longer carbon chains consume less energy in the production process than the lower alcohols, and have a shorter decomposition process for efficient heat release. Therefore, the alcohol fuel can be suitable for a mixed fuel of a diesel engine and a diesel fuel.

Many researchers have conducted some research on n-butanol fuels. The effects of biodiesel–butanol mixtures on the combustion characteristics and emissions of direct injection (DI) diesel were studied as Yilmaz et al. [14], the results show that the biodiesel–butanol mixture increases the specific fuel consumption, CO and HC emissions of the brakes compared to pure biodiesel, while reducing exhaust gas temperature and NO_x emissions. The other side of Otaka et al. [15], it is not mixed with diesel but mixed with other biofuels. The combustion characteristics and emissions of a DI engine fueled with a mixture of palm oil methyl ester and n-butanol were studied. The results show that the ignition delay of the mixed fuel is longer than that of diesel because of the low cetane number of n-butanol. Soloiu et al. [16] studied the performance and emissions of a two-cylinder engine are achieved by a combination of n-butanol port fuel injection and in-cylinder direct injection of cotton seed biodiesel for mixed combustion. From the results, it was found that a lower in-cylinder peak temperature can be obtained because of the higher endothermicity when n-butanol is vaporized. Low pressure n-butanol is injected into the intake manifold and biodiesel is injected into the cylinder. By adjusting the ratio of fuel and the injection time of biodiesel, fuel mixtures having different concentrations and reactivity can be obtained. The results show that ultra-low NO_x and soot emissions can be achieved while maintaining high efficiency.

Regarding the cetane number of n-heptane, similar to the cetane number of diesel, n-heptane is widely used as an alternative fuel to reduce the consumption of diesel fuel, mainly for exhaust gas emission experiments and kinetics of compression ignition engines in the study. In addition, it is well known that the addition of an oxidizing fuel to diesel or the addition of an oxidizing fuel as an additive can reduce hydrocarbon and soot emissions during ignition and combustion. In recent studies on exhaust emissions, many researchers have focused on the ignition and combustion characteristics of a mixture of n-heptane and an oxy-fuel.

From the perspective of chemical kinetics, some studies have been conducted to understand the decomposition and combustion of DTBP. Griffith et al., through experiments and numerical studies, compared the combustion process of DTBP in a mechanically agitated closed vessel at low pressure and a fast compressor at high pressure. According to the experimental results, the oxidation mechanism of DTBP was developed [21]. Duh et al. [22] reviewed the chemical kinetics of thermal decomposition of DTBP. Wang et al. [23] the reactivity of additives in DTBP with methanol and ethanol was studied. It was found that both exothermic and chemical effects contribute to the improvement of the reactivity of various substances, which can be attributed to the heat and activity released during the decomposition of DTBP. Chakraborty et al. [24], applying DTBP to the n-heptane decomposition process under high pressure, it was found that

DTBP accelerates the pyrolysis rate of n-heptane by the active release of organic groups after bond cleavage. Schälke et al. [25] investigated the mass burn rate of DTBP pool fire was studied, and a new model was proposed to consider the heat release rate caused due to the decomposition reaction. Sebbar et al. [26] theoretically calculated the decomposition and exothermic chemistry of DTBP, and studied DTBP in various fuels from the kinetics/decomposition, ignition and combustion of air [27].

Regarding methane, it is a good renewable gas fuel. The internal characteristics of methane have a high auto-ignition temperature of about 650°, and the flame speed is low and the range of flammability is limited. Due to its high auto-ignition temperature and low cetane number, the fuel combustion reduces particulate matter and NO_x emissions compared to pure diesel. Therefore, many research scholars have done too much research on this, such as Mathur et al. [17] compared with the CI mode, biogas is used in the dual fuel injection mode. Due to incomplete combustion of biogas during combustion, there are disadvantages of low thermal efficiency and high emissions of hydrocarbons and carbon monoxide.

In a CI engine that introduces biogas with air and diesel into the cylinder, PM emissions can be reduced and NO_x emissions can be reduced due to the high proportion of CO₂. Compared to pure diesel, biogas diesel dual fuel engines have longer ignition delays due to lower cetane numbers. This results in a shorter combustion duration resulting in incomplete combustion and reduced emissions of hydrocarbons and carbon monoxide after biofuel diesel dual fuel combustion.

In another study of biogas, in order to make the working efficiency of the dual-fuel engine reach the value equivalent to pure diesel at full load, the improved combustion can make the thermal efficiency of the engine higher than that of pure diesel, and it is necessary to increase the injection of fuel. However, it can also be changed when used with diesel fuel, which can be advanced at the beginning of the injection timing of the diesel fuel, thereby increasing the heat release rate and improving the performance.

In a compression ignition engine that reactively controls natural gas and direct injection diesel, an increase in flue gas is observed when the first diesel injection is earlier and the second diesel is injected after the piston is topped. This is because the second diesel injection occurs after the start of combustion and causes the diffusion of natural gas. Compared to diesel engines, the operation of biogas diesel engines can reduce carbon dioxide emissions because of the low carbon content of natural gas. Therefore, early work showed that by changing the carbon dioxide content in biogas, increasing the injection amount to increase the load value, reducing the flue gas emissions, and expanding the mixed value of biogas and diesel in biogas and diesel engines.

Gas can affect the combustion process in the combustion stroke. The carbon dioxide content of the biogas can be greatly reduced by physical or chemical cleaning prior to use in biogas engines. However, its focus is on determining the reduction in CO₂ content. It is important to study the effects of higher and lower CO₂ in biogas on the performance, combustion and emissions of small biogas diesel dual fuel engines. The operation of a diesel engine in which biogas is added into a dual fuel mode has been studied and observed in the prior literature, but the technology is not mature and has not been widely used in life. Yet another advanced fuel system is capable of using multiple injection strategies in biogas diesel dual fuel engines. Methane-containing biogas samples are used in common rail dual fuel engines with optimum fuel injection timing at each operating point.

Recently, in the research of DF engines, comprehensive numerical simulation and computational fluid dynamics simulation have attracted considerable attention and research in the simulation learning of engines. Interpretation using a

set simulation model DF combustion ignition is used as a volumetric process combined with three ignition phases: (1) low temperature ignition combustion simulation, (2) high temperature ignition combustion simulation, and (3) premixed methane-air Flame combustion simulation was initiated in the mixture. In addition, methane fuel was found to initiate an inhibitory reaction by consuming an OH group with less cetane number of itself, delaying the early decomposition of the pilot fuel.

Through the study of numerical studies, many researchers have experimentally studied the simulation of the DF combustion process in a single or multi-cylinder engine.

Alla et al. [18] studied the effect of single-cylinder engine on the combustion performance of diesel–methane mixed fuel DF under lean conditions. The results show that NO_x emissions increase due to high temperature combustion in the cylinder. The use of a large number of pilot diesel injections improves the combustion process of the engine and promotes full combustion of the fuel, thus reducing CO and UHC emissions. In addition, under high load, as the amount of pilot diesel increases, the increase in combustion duration and the occurrence of knock occurs earlier. Papagiannakis et al. [19] studied the effects of NG on DF combustion under three different engine loads. The results show that as the natural gas content increases, the ignition delay time increases and the combustion duration decreases, which promotes the complete mixing rate and reduces exhaust emissions. However, UHC emissions are still high under low load conditions, so, gaseous fuels can significantly affect combustion. In addition, the physical and chemical interactions between the pilot fuel and the gaseous fuel play an important role in affecting the overall combustion performance throughout the combustion process.

Beside these researchers' research, a lot of researchers have investigated the dual-fuel injection in diesel engine, and got some effective results.

Abu-Qudais et al. [9] studied that the combustion characteristics of different proportions of ethanol–diesel blended fuel inside the engine were tested in a compression ignition internal combustion engine. In this study, by using a volume of 15% and 20% ethanol as the test fuel for the engine, the particulate matter emissions after combustion were reduced by 32% and 51%, respectively. This is also studied in Karabektas et al. [10]. Iso-butanol is mixed with diesel at different ratios. And from the results found, as the volume ratio of the iso-butanol content in the fuel mixture increases, the specific fuel consumption and thermal efficiency increase. The CO and NO_x emissions in the blended fuel increase. Since, the increase in iso-butanol is a decrease in the cetane number of combustion, the ignition delay is increased and the combustion process is reduced and causes a decrease in CO. However, the higher endo thermicity of iso-butanol leads to a reduction in NO_x emissions. Therefore, the use of iso-butanol-diesel blends reduces CO and NO_x emissions, while HC emissions increase. Singal et al. [11] studied the effects of fuel characteristics on emissions. From this study, it is shown that sulfur content and fuel density have an important impact on particle emissions. And this result is also proved from other studies. Tsurutani et al. [12] studied the characteristics of fuels that play an essential role in the combustion process, such as the effect of oxidative combustion of fuel on emission characteristics. It is shown from the results that the amount of the aromatic diameter can cause an increase or decrease in the discharge of the particles. Yoshiyuki et al. studied the effects of fuel properties on diesel combustion and emissions. It was found from the results that lowering the cetane number can lead to an increase in NO_x emissions and a decrease in soot. Akasaka et al. [13] studied the effects of fragrant hydrocarbon content on combustion and emission characteristics from the perspective of fuel combustion chemistry. Therefore, it was found that the molecular structure of fuel plays an important role in the generation of particulate

emissions.

The main focus of this paper is to study the combustion characteristics of fuels and the critical reactions to the combustion characteristics of different fuels and their relationship to combustion characteristics under certain piston shapes. Different fuel injections, the distribution of combustion during combustion, the formation of nitrogen oxides during combustion, and the distribution of temperature in a particular piston shape.

Simulation Basic Setting

In this work, the fluid analysis software ANSYS was used for model simulations related to fuel spray and combustion. The method of calculating and simulating the generation of turbulence is achieved by the Reynolds average Navier–Stokes (RANS) method, in which the instantaneous analog quantity is decomposed into time-averaged quantities and fluctuations. The spray wall interaction model uses a rebound/sliding model. A turbulence model with an enhanced RNGk- ϵ model is used.

The turbulence model used in the simulation is mainly the (RNG) k- ϵ model. It has been proved by the past research that the RNG model simulation analysis can be applied to the analysis of complex turbulent flow fields, and the error can be effectively reduced in the simulation, and better results can be obtained in numerical calculation. The model can also be used to calculate the simulation of compressible turbulence, which is usually represented by the kinetic energy equation k and the dissipation rate equation ϵ during the simulation shown as:

$$\frac{D(\rho k)}{Dt} = P - \rho \epsilon + D_k \quad (1)$$

$$\frac{D(\rho \epsilon)}{Dt} = C_1 \frac{P \epsilon}{k} - C_2 \rho \frac{\epsilon^2}{k} + C_3 \rho \epsilon (\nabla \cdot \mathbf{u}) + D_\epsilon \quad (2)$$

+ D where, $\mathbf{u}$ is the average velocity vector, P is the turbulent energy generation term, and D_k and D_ϵ are the turbulent diffusion terms.

Condition and Model

In this simulation paper, we mainly study the distribution of temperature during combustion of nitrogen oxides and fuels in a specific environment after injection of different fuels. Therefore, the model studied is a direct injection engine with a bore diameter of 85.5 mm and a compression ratio of 14.7:1. The main focus is to simulate the combustion of the piston in a certain cylinder environment, so, the process of exhaust is not considered, so this simulation study does not involve intake and exhaust valves. The interval of the simulation depends on the moment when the intake valve is closed and the exhaust is about to open. In order to make the simulation more reliable and close to the actual working state of the diesel engine, the simulation considers that the cylinder contains a small amount of residual exhaust gas, so the intake valve is closed. The substance in the cylinder is defined as fresh air and the exhaust gas retained in the previous cycle, as well as the injected fuel. The fuel ratio is less than 1, which can be called as lean combustion. The selection of this model is selected according to the ratio of 1:1 to the real model, and the model is established according to 2D. This model mainly sets the total number of grids set to 30,696 under the condition of compression combustion after a specific intake valve is closed. The geometric model can be as shown in figure 1.

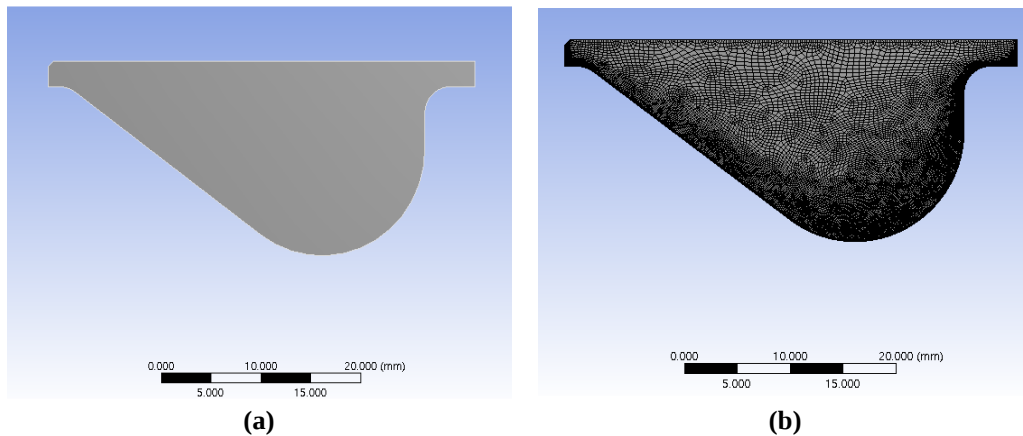


Figure 1: Piston Geometry.

In this paper, a two-dimensional model is adopted. The turbulence model selects the standard k-epsilon model. By changing the fuel injected, the fuel is suitable for the engine from different combustion characteristics. However, the boundary conditions and initial conditions of the numerical simulation are prerequisite simulations to ensure that the numerical calculation has a solution. The boundary conditions of this model mainly refer to the physical boundary conditions, as well as the chemical properties of the fuel and the physical properties of the fuel itself. We can refer to the characteristics of various fuels as shown in table 1.

Table 1: The Fuel Characteristics

Properties	Unit	Diesel	Toluene	N-Butanol	N-Heptane	Methane
Density	gm/ml	0.85	0.87	0.81	0.692	0.657
Flash Point	°C	55	6	34	25	-188
Viscosity	cSt	3.0	0.59	2.9	0.42	17.8
Calorific Value	MJ/kg	42	40.59	33.1	44.5	50
Ignition Temperature	°C	250	480	343	371	537
Cetane Number	-	45	7.4	~25	56	0

From table 1, we can get a preliminary understanding of the physical and chemical properties of each fuel. The nature of the fuel plays an important role in the combustion of the internal combustion engine and the exhaust gas. From the table we can find the density characteristics and the viscosity of the fuel. From the surface phenomenon, we can preliminarily conclude that N-heptane can be effectively burned with air and can be fully burned, depending on the density and viscosity of the fuel. Another characteristic that affects fuel combustion is the cetane number. The cetane number plays an important role in the fuel's auto-ignition time delay. The fuel with a larger cetane number causes a shortening of the auto-ignition delay. N-heptane is a suitable fuel for internal combustion engines. Cetane number also has a significant effect on tail gas emissions. A fuel with a large cetane number causes an increase in combustion and shortens the combustion duration time. At the end of combustion, the temperature in the cylinder is low, so that NO_x emissions tend to decrease.

RESULTS AND DISCUSSIONS

Fuel Injection Amount Distribution

Figure 2 shows us and compare the different fuels distributions inside the cylinder. We can see the direction and degree of mixing of the fuel from the figure. For a certain amount of fuel injection, the distribution of n-heptane is relatively uniform, mainly depending on the viscosity characteristics of the fuel itself. The viscosity of n-heptane is relatively low (Table 1).

However, due to this viscosity characteristic, it is easy to find that the distribution of methane fuel is not uniform, resulting in uneven combustion. The degree of evaporation of the fuel is also affected by the viscosity of the fuel. The fuel with a small viscosity value causes high volatility of the fuel, thereby promoting the mixing of air and fuel, and promoting the completeness of the fuel.

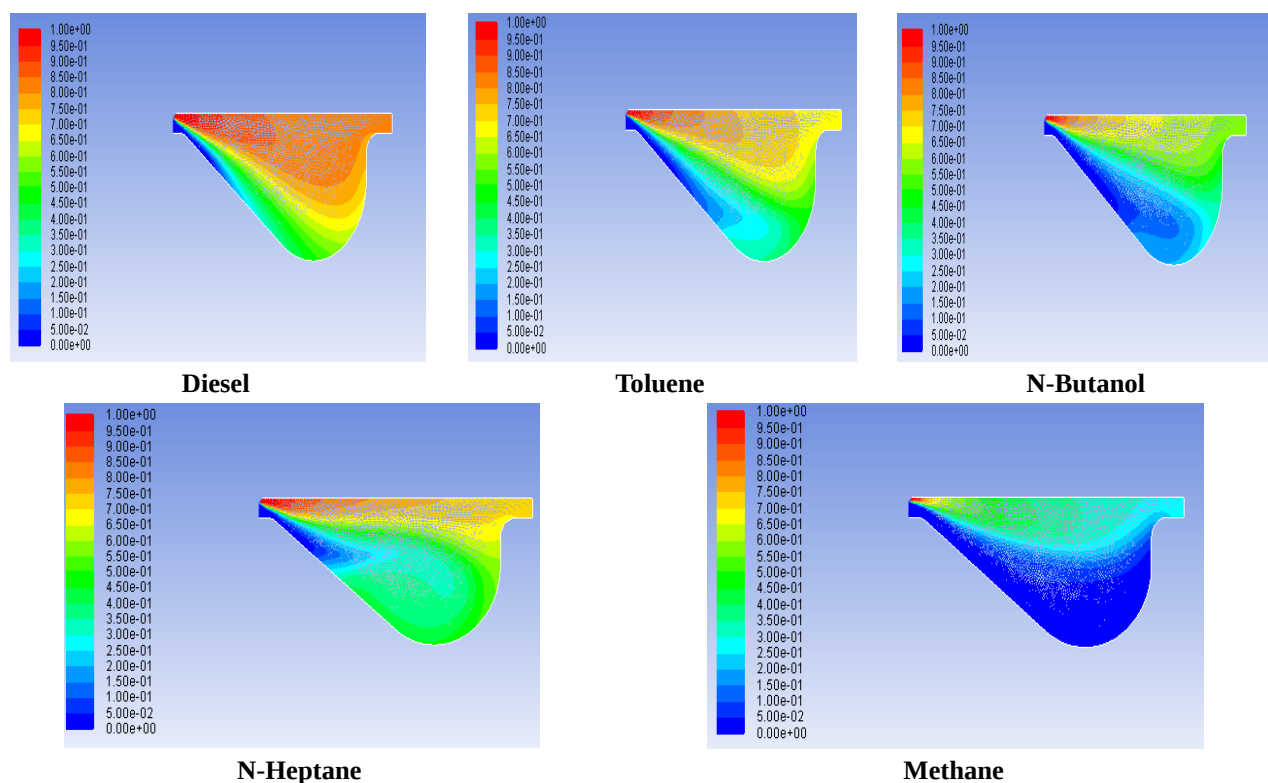


Figure 2: Fuel Distributions in Cylinder.

Combustion Characteristics

We can see the temperature distribution in the cylinder when the fuel is burned from figure 3. However, the difference in the cetane number of each fuel leads to the difference in the auto-ignition time of the fuel. From the surface map of the temperature distribution, we can observe the cylinder after the fuel is burned. The maximum temperature inside and the auto-ignition delay time, but related to the specific temperature distribution and time, is a bit less clear, so we can observe it in the graph below. It is apparent that the position of each fuel at the time of combustion and the difference in the maximum temperature value reached in the cylinder are found. The appearance of this phenomenon depends mainly on the peculiar nature of the fuel itself. The difference in combustion position (ie, the self-ignition delay time is different) depends on the cetane number of the fuel and the cetane number (diesel and N-heptane). The self-ignition delay is relatively small, compared to a relatively low cetane number (especially Methane) combustion delay. This caused a high temperature in the later stages of combustion, which caused an increase in NO_x emissions. It can be seen from Figure 3 that fuels with maximum temperature include diesel and Methane, but have different effects on the engine due to different burning times. For the other three fuels, N-heptane has a relatively low maximum temperature value, which should be due to its own density and calorific value.

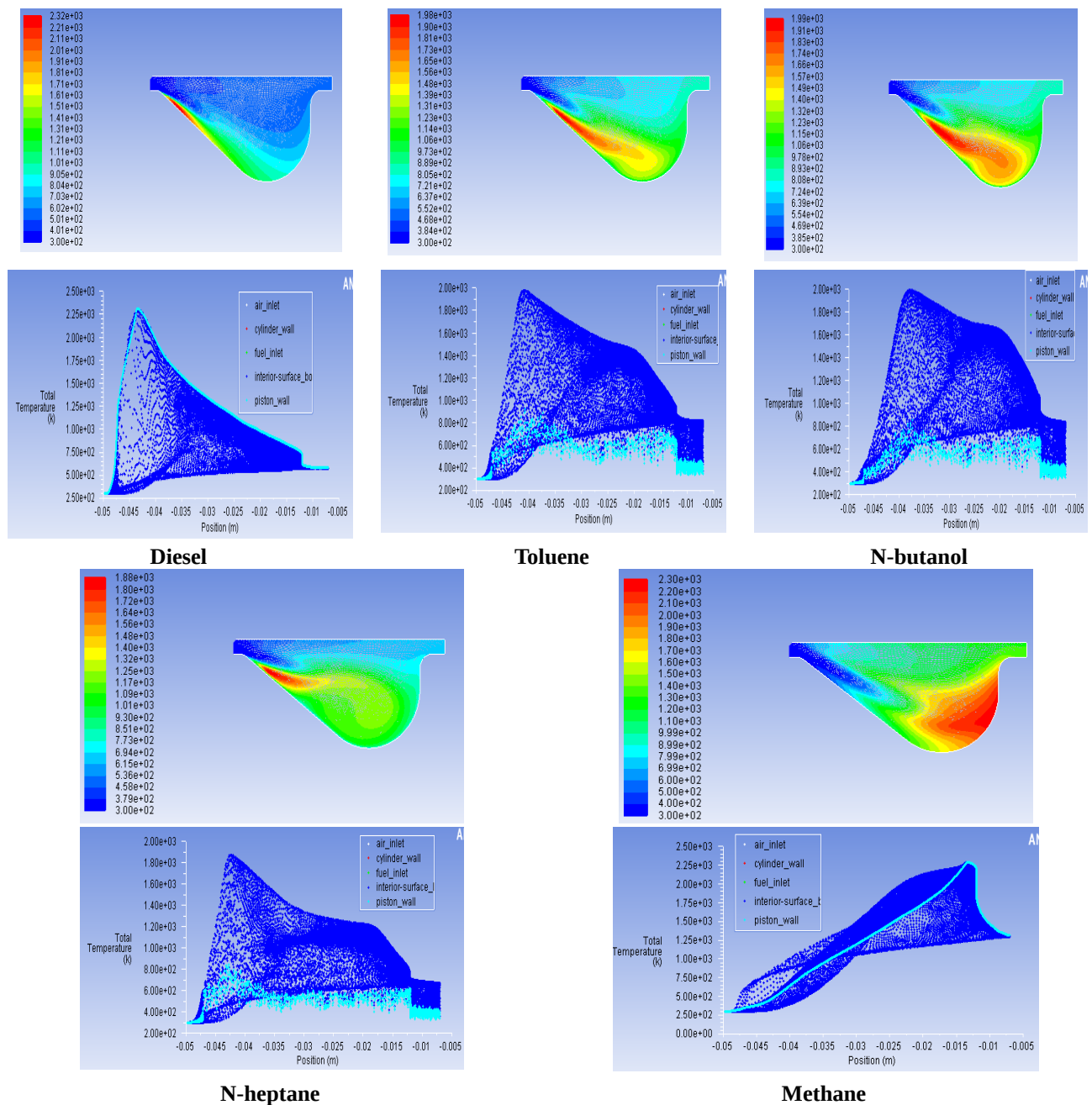


Figure 3: The Total Temperature of the Combustion.

Turbulent Rate of Reaction with Various Fuels

Figure 4 shows the turbulent flow rate of different fuel injections in the cylinder. The turbulent flow rate distribution can be obtained from the figure. The order of turbulence rate from large to small is N-heptane-Diesel-Toluene-N-butanol-Methane. One side, maybe the fuel's properties and the mixture rate of fuel and air cause the turbulentant happened. The viscosity of fuel cause the combustion delay and make the fuel combustion velocity different because the evaporation causes fuel and air complete mixture causes the combustion temperature different, thus causes different turbulence in the fuel inside the cylinder. Another side maybe the different cetane number of the fuel inside cylinder, causes the ignition delay, high cetane

number cause the ignition delay short, make the combustion happened early caused the internal temperature of the cylinder to rise quickly, high temperature make the more turbulence occurred (Table 1). This effects maybe caused by different fuel chain. The way in which the base chains in the fuel are connected, as well as the ease with which the branches break, can also affect the rate of combustion. The larger the carbon number, the longer the chain, the easier it is to break, and the faster the burning rate.

Figure 4 show us the N-heptane's turbulent rate is higher in the injected fuels and the Methane's turbulent rate is lower than other fuels, because its property has high cetane number and low viscosity. Through this comparison in Figure 4, shows us the nearby the cylinder wall and piston wall turbulent effects.

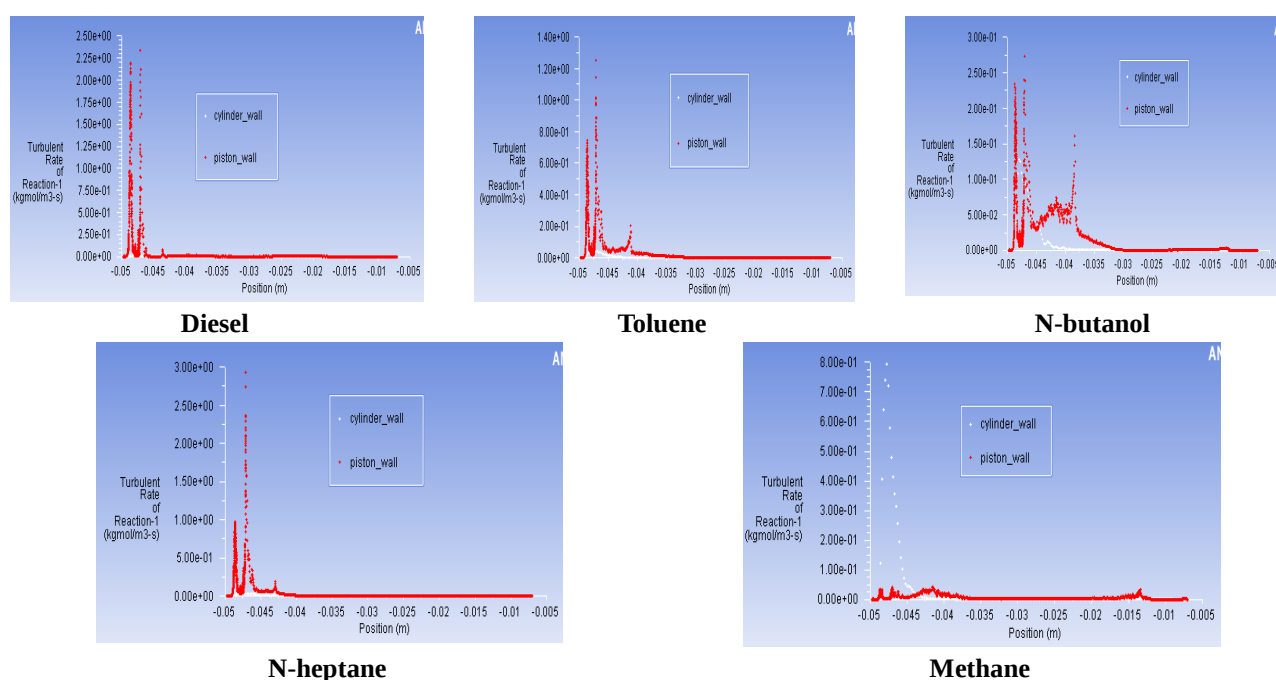


Figure 4: Turbulent Rate of Reaction with Various Fuels.

Emissions Analysis

Through above analysis fuel physical and chemical characteristic, we found different fuels cause the cylinder temperature, combustion distributions, fuel turbulent rate, beside, cylinder wall and piston wall. In this paper, we will check the emissions after the fuel combustion, Specially NO_x and SO₂ emissions. Through Figures 5 and 6, we can check the exhaust emissions.

It can be seen from the figures 4 and 5 that the relationship between turbulence and NO_x emission, the turbulent flow of toluene and n-butanol in the cylinder is large, resulting in better mixing of the fuel in the cylinder, and the fuel reaches full combustion of its own. The lower cetane number causes a delay in combustion, which results in a longer combustion time, which is a higher temperature in the cylinder and affects more nitrogen oxide emissions.

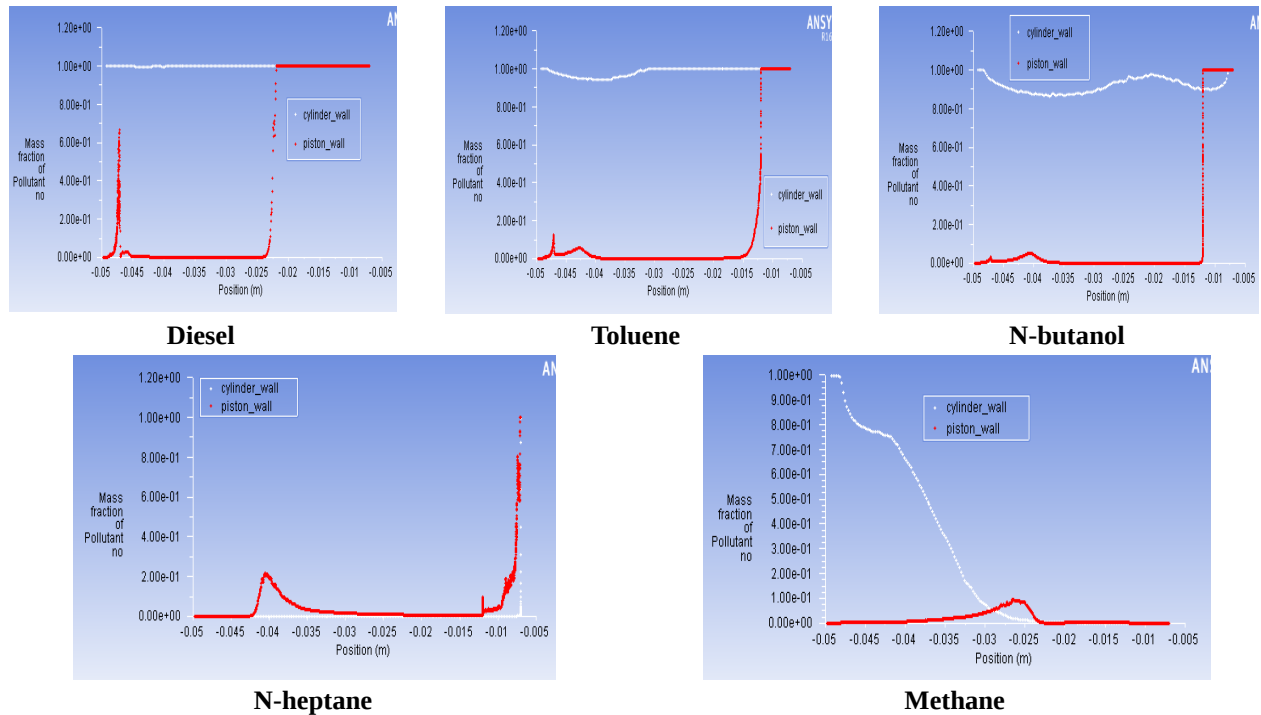
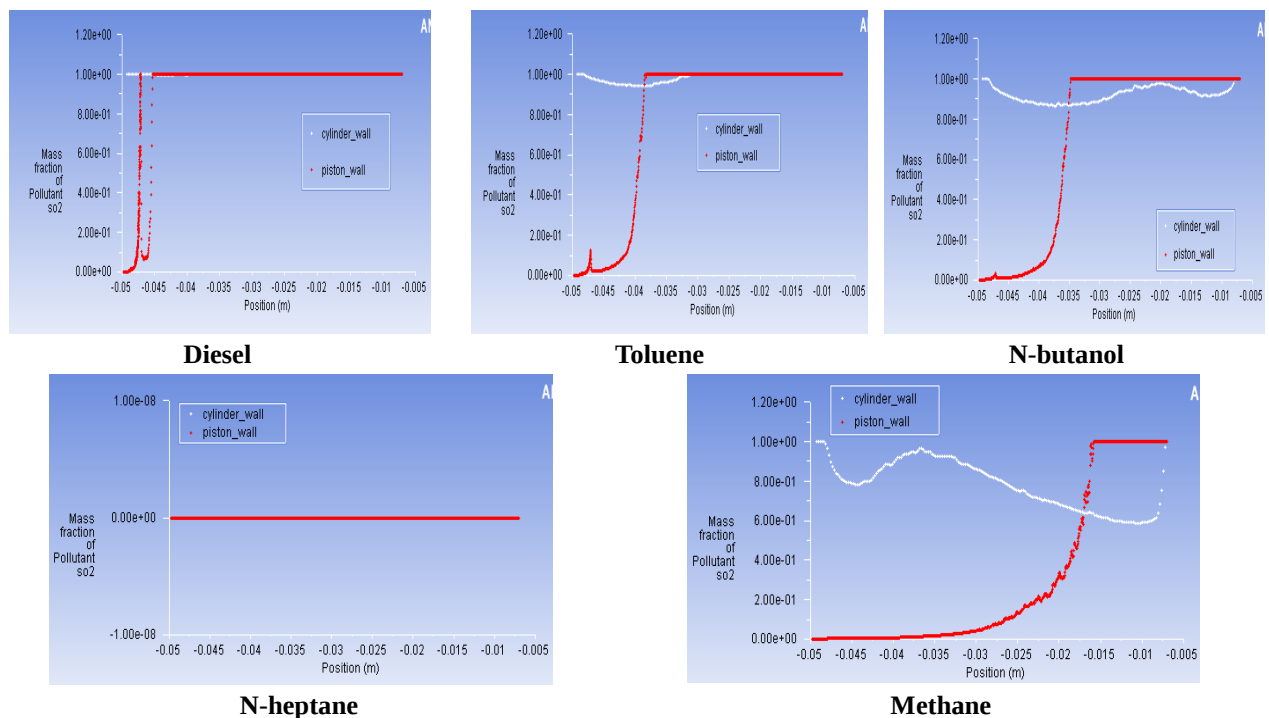


Figure 5: NO Emissions with Various Fuels.

Figure 6: SO₂ Emissions with Various Fuels.

Figures 5 and 6 analyze the exhaust gas conditions produced by the combustion of the fuel. Here, we mainly study the emissions of nitrides, sulfides, and soot. To make the data obvious, we analyze the curve direction of the cylinder boundary (cylinder wall). And from the piston wall, we can also regard the part between the two curves as the exhaust gas distribution inside the cylinder. This can clearly show the curve distribution.

It is found from the figure that N-heptane is the least produced by nitrogen oxides, followed by Methane, and the most is diesel fuel. However, the main reason for the generation of nitrogen oxides is the state of their in-cylinder temperature. We can refer to Figure 3.

Figure 6 shows the distribution of sulfide formation. It is found that N-heptane does not produce sulfides at all, while other fuels have sulfide formation. Sulfur compounds from diesel fuels are the most prominent. After combustion, it tends to be stable. Methane's combustion process is longer, mainly because it contains the smallest cetane number, causing combustion delays that affect the combustion process.

CONCLUSIONS

For different fuels, the nature of the fuel plays a major role in the combustion process of the internal combustion engine. The fuel contains various properties including cetane number, viscosity value and temperature of spontaneous combustion.

Among them, the cetane number is the most prominent. The size of the cetane number can affect the length of time and makes the fuel spontaneous combustion delay. Thereby, affect the temperature of combustion in the cylinder, affecting the emission of nitrogen oxides.

The viscosity value affects the flow rate of the fuel and the rate of evaporation, thereby affecting the complete combustion of the fuel and air mixture.

The results show that the calculated value of the in-cylinder temperature verifies the correctness of the model. From the simulation results, it can be found that the total temperature in the n-heptane cylinder, the NO_x least and the turbulent rate the largest. However, the NO_x and SO₂ active groups produced by the combustion reaction in the n-heptane cylinder are the least.

It can be seen from the simulation of this paper that the combustion performance of n-heptane is close to that of diesel, and in comparison, less polluted exhaust gas is produced. It is ideal to mix with diesel to make biofuels or as an alternative fuel for diesel.

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REFERENCES

1. LI Rui-na, WANG Zhong*, LI Ming-di, ZHAO Yang, NI Pei-yong, LIU Shuai, *Experimental study of cetane number on methanol/biodiesel emission of pollutant*. *China Environmental Science*, 2014, 34(7) : 1684–1689.
2. Seyed Reza Amini Niaki, Sajad Mahdavi, and Joseph Mouallem, *Experimental and Simulation Investigation of Effect of Biodiesel-Diesel Blend on Performance, Combustion, and Emission Characteristics of a Diesel Engine*. *Environmental Progress & Sustainable Energy*.
3. V.C. Momale, S.A. Giri, M.S. Yadav, *Compression Ignition Engine Simulation using Computational Fluid Dynamics*, *International Advanced Research Journal in Science, Engineering and Technology* Vol. 3, Issue 6, June 2016.
4. Gianluca D'Errico, Tommaso Lucchini, Frank Atzler, and Rossella Rotondi. *Computational Fluid Dynamics Simulation of Diesel Engines with Sophisticated Injection Strategies for In-Cylinder Pollutant Controls*. *Energy Fuels* 2012, 26, 4212–4223.
5. Fan Qianwang, Liliguang, Deng Jun, HU, Zongjie, *Effect of various Fuels on combustion and key species for compression*

- ignition engines. *Journal of Tongji University (Natural science)* 06.2016.vol.39.
6. H. Barths, H. Pitsch, N. Peters. *Three-Dimensional Simulation of DI Diesel Combustion and Pollutant Formation Using a Two-Component Reference Fuel.*
 7. Md.Nasir Uddin, M.M Rashid, N.A Nithe, J.I Rony. *An Investigation on the Performance of a Single Cylinder Diesel Engine Using Biodiesel, Proceedings of the International Conference Biotechnology Engineering, July 25-27, 2016.*
 8. Kandasamy, A., Jabbaraj, D., & Chandran, M. *Investigation of Biodiesel (Jatropha) with AL₂O₃ in CI Engine.*
 9. Yahuza RI, Ejilal, Dandakouta H and Farinwata SS, *Exhaust Emissions Characterization of a Single Cylinder Diesel Engine Fueled with Biodiesel-Ethanol-Diesel Blends, Bioinformatics & Proteomics Open Access Journal. Volume 2 issue 1.*
 10. Abu-Qudais M, Haddad O, Qudaisat M. *The effect of alcohol fumigation on diesel engine performance and emissions. Energy Convers Manage 2000; 41(6):389–99.*
 11. Karabektaş M, Hoşöz M. *Performance and emission characteristics of a diesel engine using isobutanol-diesel fuel blends. Renewable Energy 2009; 34(6):1554–9.*
 12. Singal S K, Singh I P, *Fuel quality requirements for reduction of diesel emission[C] SAE 1999 world congress & Exhibition. SAE, 1999, 1999-01-3592.*
 13. Tsurutani K, Takei Y, Fujimoto Y, et al. *The effect of fuel properties and oxygenates on diesel characteristic of a direct injection diesel engine, SAE 2000 world congress & Exhibition. 2000-01-1851.*
 14. Akasaka Y, Sakurai Y, *Effect of fuel properties on exhaust emission from DI diesel engine. JSME International Journal, Series B, 1988, 41(1):13.*
 15. Yilmaz N, Vigil FM, Benalil K, Davis SM, Calva A. *Fuel 2014; 135:46–50.*
 16. Otaka T, Fushimi K, Kinoshita E, Yoshimoto Y. *SAE Technical Paper 2014-32-0085 2014.*
 17. Soloiu V, Duggan M, Ochieng H, Harp S, Weaver J, Jenkins C, et al. *SAE Technical Paper 2013-01-0916 2013.*
 18. Mathur HB, Gajendra Babu MK, Prasad YN. *A thermodynamic simulation model for a dual fuel open combustion chamber compression ignition engine. SAE Technical Paper 861275, 1986.*
 19. Sanju, D., Prabakaran, B., & Vijayabalan, P. *Influence of Addition of Ethanol into Non-Edible MOBD an Experimental Study on the Performance Of CI Engine.*
 20. Abd Alla GH, Soliman HA, Badr OA, Abd Rabbo MF. *Effect of pilot fuel quantity on the performance of a dual fuel engine. Energy Convers Manage 2000;41(6):559–72.*
 21. Papagiannakis RG, Hountalas DT. *Combustion and exhaust emission characteristics of a dual fuel compression ignition engine operated with pilot diesel fuel and natural gas. Energy Convers Manage 2004; 45(18–19):2971–87.*
 22. Shan X, Qian Y, Zhu L, Lu X. *Effects of EGR rate and hydrogen/carbon monoxide ratio on combustion and emission characteristics of biogas/diesel dual fuel combustion engine. Fuel 2016; 181:1050–7.*
 23. Griffiths JF, Phillips CH. *Experimental and numerical studies of the combustion of di tertiary butyl peroxide in the presence of oxygen at low pressures in a mechanically stirred closed vessel. Combust Flame 1990; 81:304–16.*
 24. Duh Y-S, Kao C-S, Lee W-L. *Chemical kinetics on thermal decompositions of di-tertbutyl peroxide studied by calorimetry. J Therm Anal Calorim 2017; 127:1071–87.*
 25. Wang H, Dempsey AB, Yao MF, Jia M, Rolf RD. *Kinetic and numerical study on the effects of di-tert-butyl peroxide additive on*

the reactivity of methanol and ethanol. *Energy Fuels* 2014; 28:5480–8.

26. Chakraborty JP, Kunzru D. High-pressure pyrolysis of n-heptane: effect of initiators. *J Anal Appl Pyrol* 2012; 95:48–55.
27. Schälike S, Chun H, Mishra KB, Wehrstedt KD, Schönbucher A. Mass burning rates of di-tert-butyl peroxide pool fires—experimental study and modeling. *Combust SciTech* 2013; 185:408–19.
28. Sebban N, Bozzelli JW, Bockhorn H. Kinetic study of di-tert-butyl peroxide: thermal decomposition and product reaction pathways. *Int J Chem Kinet* 2015; 47:133–61.
29. Naik, R. T., & Nilesh, C. (2016). Emission characteristic of a high speed diesel engine. *International Journal of Mechanical Engineering*, 5, 29–36.
30. Sebban N, Habisreuther P, Bockhorn H, Auzmendi-Murua I, Bozzelli JW. Di-tertiarybutylperoxide decomposition and combustion with air: reaction mechanism, ignition, flame structures, and laminar flame velocities. *Energy Fuels* 2017; 31:2260–73.

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